US007451617B2

(12) **United States Patent**
Renz et al.

(10) **Patent No.: US 7,451,617 B2**
(45) **Date of Patent: Nov. 18, 2008**

(54) **REFRIGERATION SYSTEM**

(75) Inventors: **Hermann Renz**, Egenhausen (DE); **Günter Dittrich**, Oschatz (DE)

(73) Assignee: **Bitzer Kuhlmaschinenbau GmbH** (DE)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **11/840,344**

(22) Filed: **Aug. 17, 2007**

(65) **Prior Publication Data**

US 2008/0011014 A1 Jan. 17, 2008

Related U.S. Application Data

(63) Continuation of application No. PCT/EP2006/000581, filed on Jan. 24, 2006.

(30) **Foreign Application Priority Data**

Feb. 17, 2005 (DE) 10 2005 009 173

(51) **Int. Cl.**
F25B 1/10 (2006.01)

(52) **U.S. Cl.** **62/510**; 62/228.5

(58) **Field of Classification Search** 62/228.5, 62/509, 510, 511, 513, 515
See application file for complete search history.

(56) **References Cited**

U.S. PATENT DOCUMENTS

4,197,719 A 4/1980 Shaw
4,254,637 A * 3/1981 Brauch et al. 62/468
5,522,233 A * 6/1996 Nares et al. 62/193
6,105,378 A 8/2000 Shaw
6,145,326 A * 11/2000 O'Brien 62/84

(Continued)

FOREIGN PATENT DOCUMENTS

CH 227856 7/1943

(Continued)

OTHER PUBLICATIONS

Ian Bell, Performance Increase of carbon dioxide Refrigeration Cycle with the Addition of Parallel Compression Economization, paper, 9 pages, XP-000962591, 2/A/4.30.

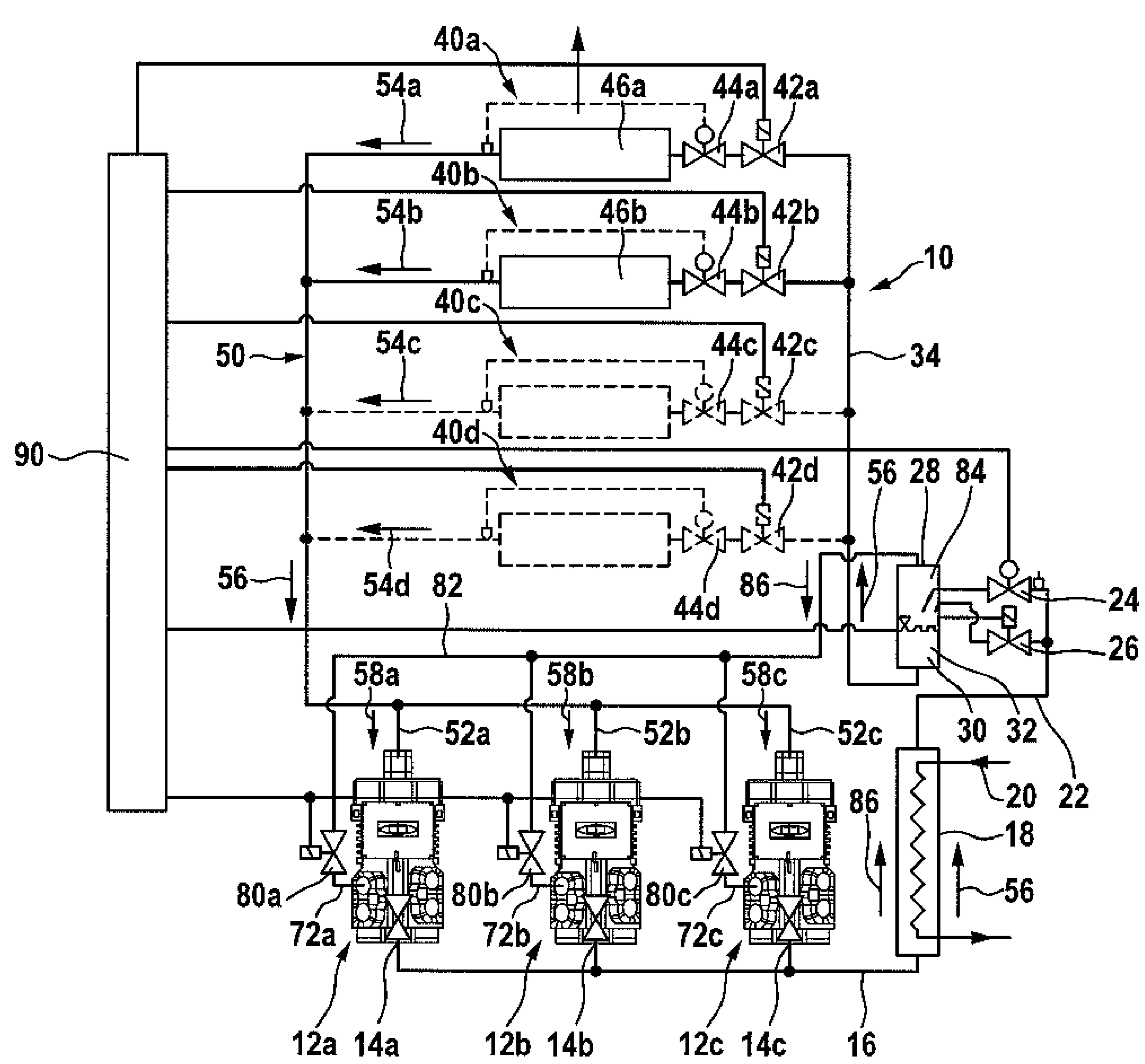
(Continued)

Primary Examiner—Melvin Jones
(74) *Attorney, Agent, or Firm*—Reinhart Boerner Van Deuren P.C.

(57) **ABSTRACT**

In order to create a refrigeration system comprising a refrigerant circuit (**10**), in which a main mass flow (**56**) of a refrigerant is guided, which may be adapted to different operating conditions in an optimum manner it is suggested that at least two refrigerant compressors (**12***a*, **12***b*, **12***c*), which can be switched on individually for the purpose of compressing the main mass flow, be arranged in the refrigerant circuit, that at least two of the refrigerant compressors each have at least one additional compressor stage (**70**), that each of the additional compressor stages be able to be used optionally for the compression of refrigerant from the main mass flow or for the compression of refrigerant from the additional mass flow (**86**) and that a control (**90**) be provided, with which such a number of additional compressor stages for the compression of refrigerant from the additional mass flow can be switched on in a first operating mode as a function of the operation conditions that the expansion cooling device (**24**, **28**) liquefies the main mass flow and reduces its enthalpy.

28 Claims, 10 Drawing Sheets



U.S. PATENT DOCUMENTS

6,931,871	B2 *	8/2005	Shaw et al.	62/228.5

FOREIGN PATENT DOCUMENTS

DE	3925090	A1	2/1991
DE	4309137	A1	8/1994
EP	0180904	A2	5/1986
EP	0845642	A2	6/1998
EP	1357338	A1	10/2003
EP	1498667	A2	1/2005
GB	2199931	A	7/1988
WO	WO 2006/015741	A1	2/2006

OTHER PUBLICATIONS

Sergio Girotto et al., Commercial refrigeration system using CO2 as the refrigerant, journal, 2004, International Journal of Refrigeration 27 (2004), 7 pages, 717-723.

Cosijn E A, Toepassing van tweetraps (compound-) zuigercompressoren in de industriële koeltechniek, Nederlandse Vereniging Voor Koeltechniek, Apeldoorn, NL, vol. 82, No. 5, May 1, 1989, pp. 24-25, XP000022690.

* cited by examiner

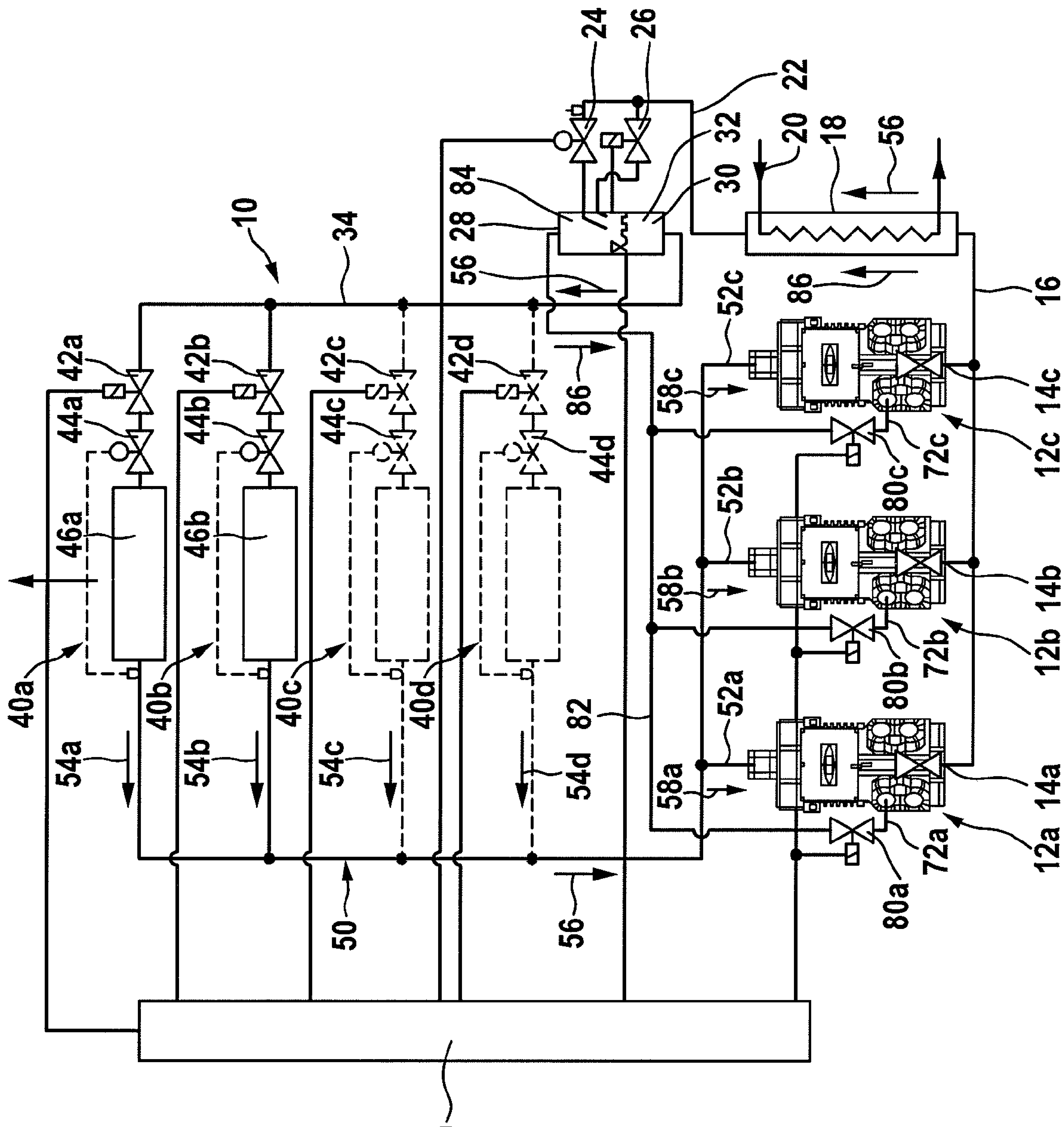

10
34
24
26
22
84
28
32
30
20
18
56
86
16
52c
52b
52a
58c
58b
58a
14c
14b
14a
12c
12b
12a
72c
72b
72a
80c
80b
80a
82
42a
42b
42c
42d
44a
44b
44c
44d
46a
46b
40a
40b
40c
40d
54a
54b
54c
54d
50
90

Fig. 1

Fig. 2

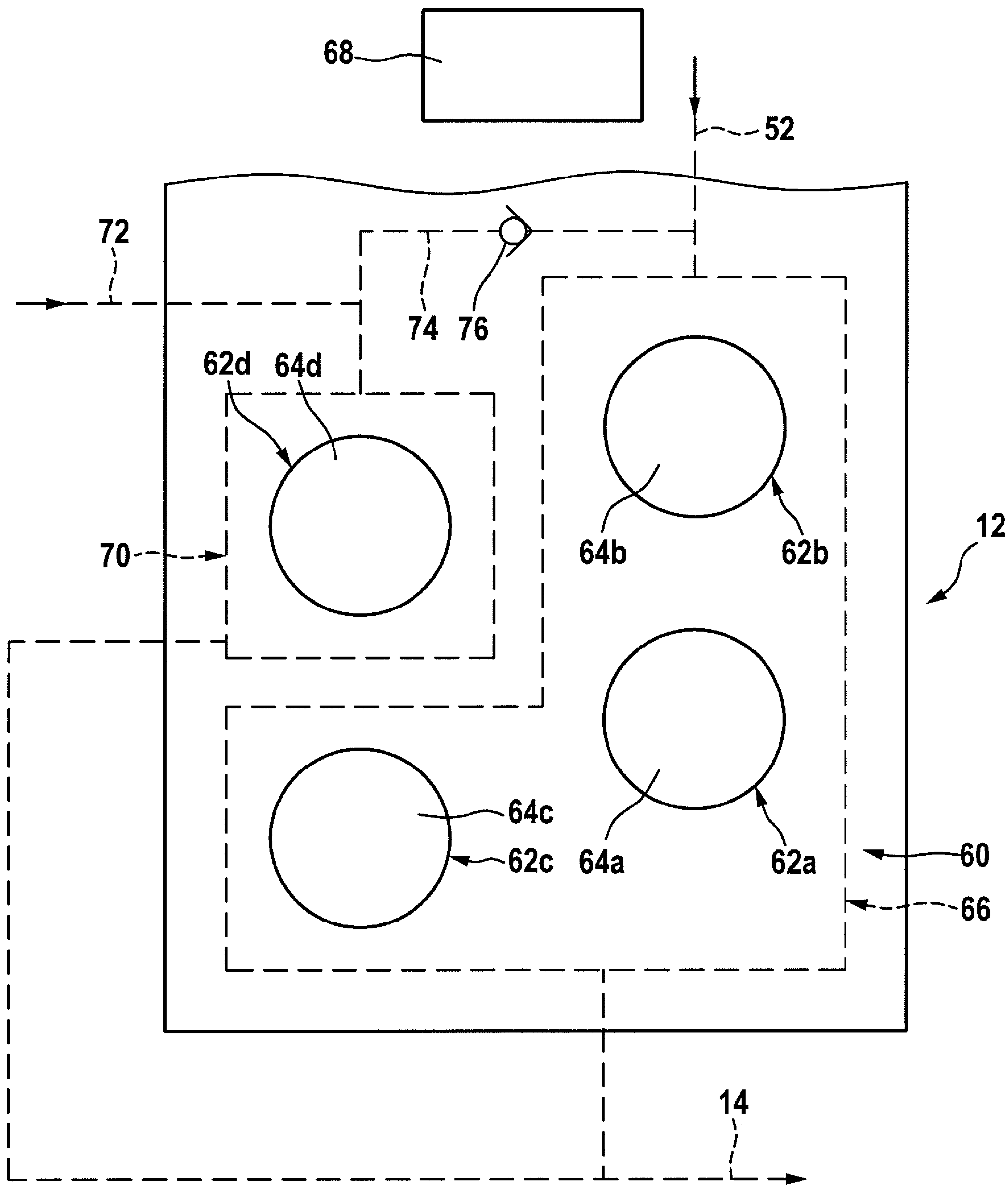
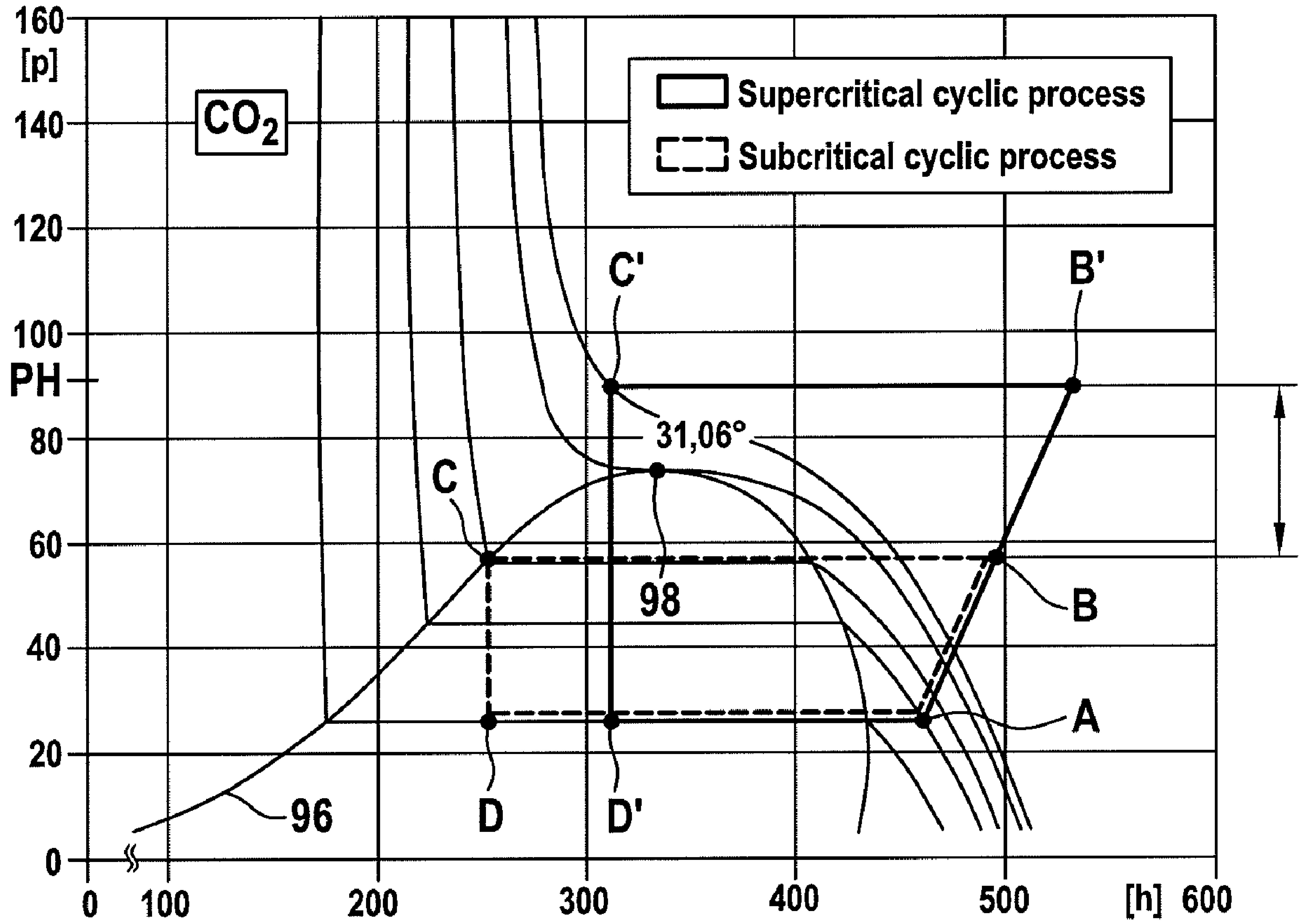


Fig. 3



160
[p]
140
120
100
PH
80
60
40
20
0
0
100
200
300
400
500
[h]
600
CO2
Supercritical cyclic process
Subcritical cyclic process
C'
B'
31,06°
C
98
B
A
96
D
D'

Fig. 4

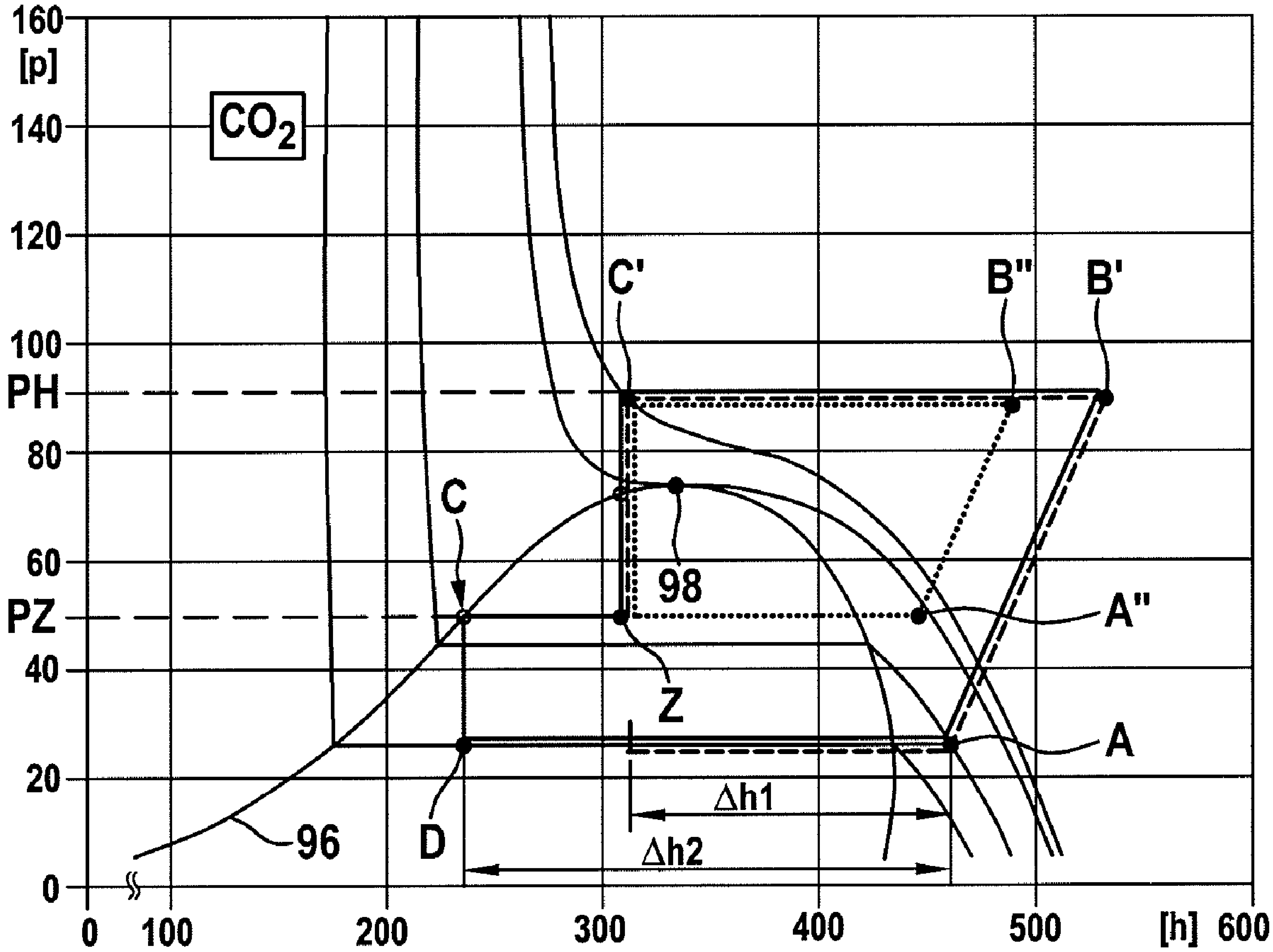

160
[p]
140
120
100
PH
80
60
PZ
40
20
0
CO_2
C'
B"
B'
C
98
A"
Z
A
96
D
Δh1
Δh2
0
100
200
300
400
500
[h]
600

Fig. 5

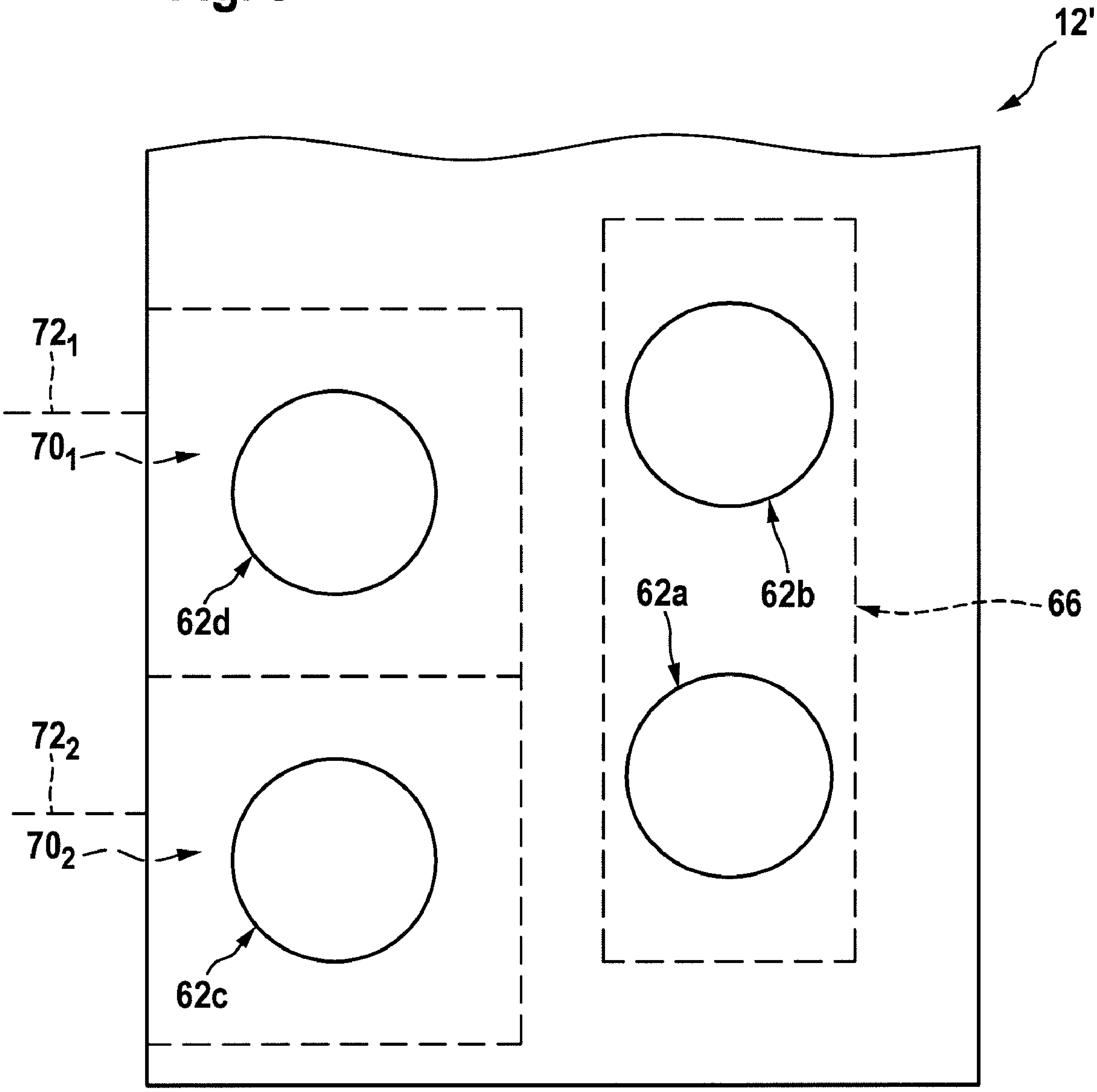
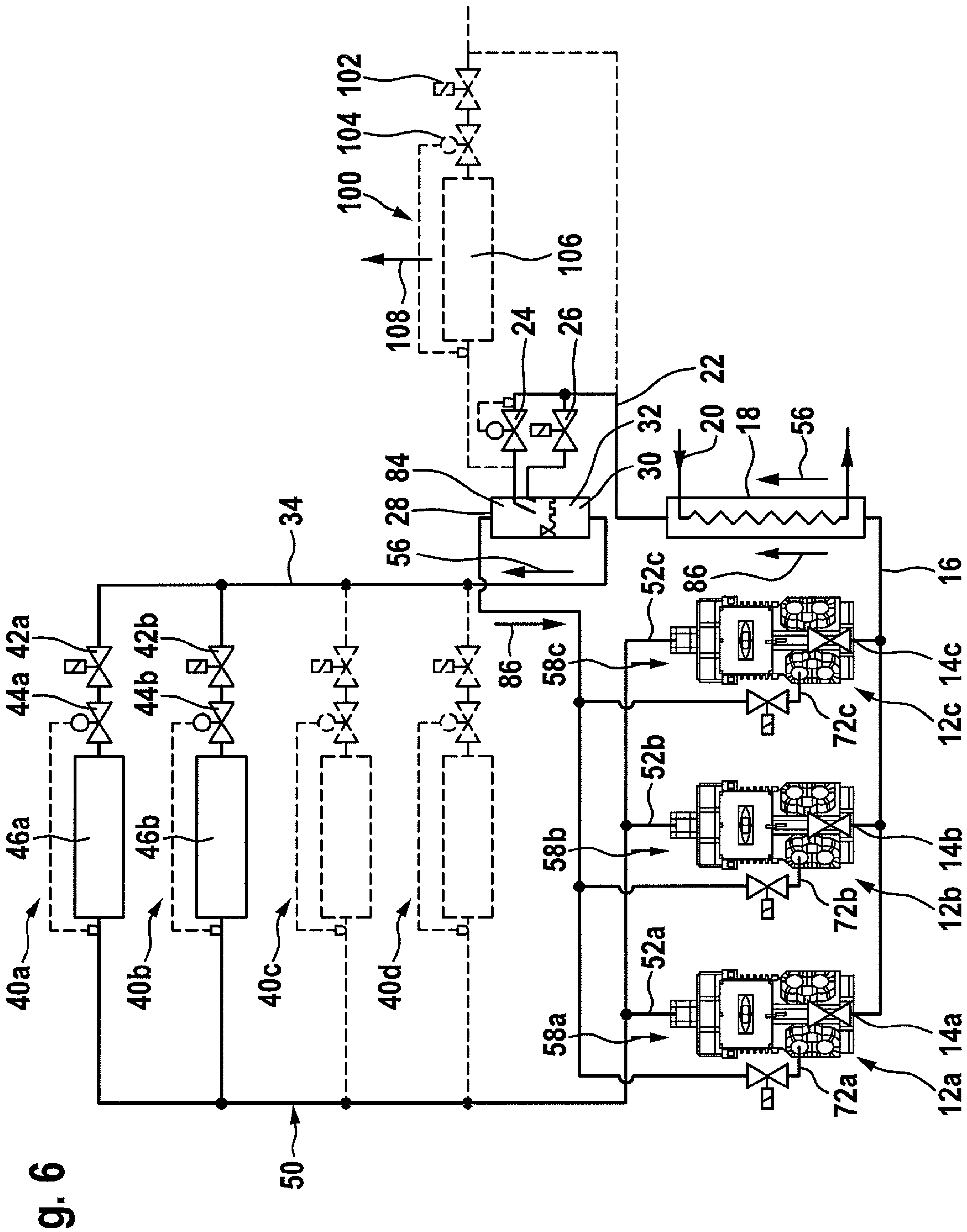


Fig. 6



102
104
100
106
108
24
26
22
84
28
32
30
20
18
56
34
56
86
52c
86
58c
14c
12c
72c
52b
58b
14b
12b
72b
52a
58a
14a
12a
72a
16
40a
46a
44a
42a
40b
46b
44b
42b
40c
40d
50

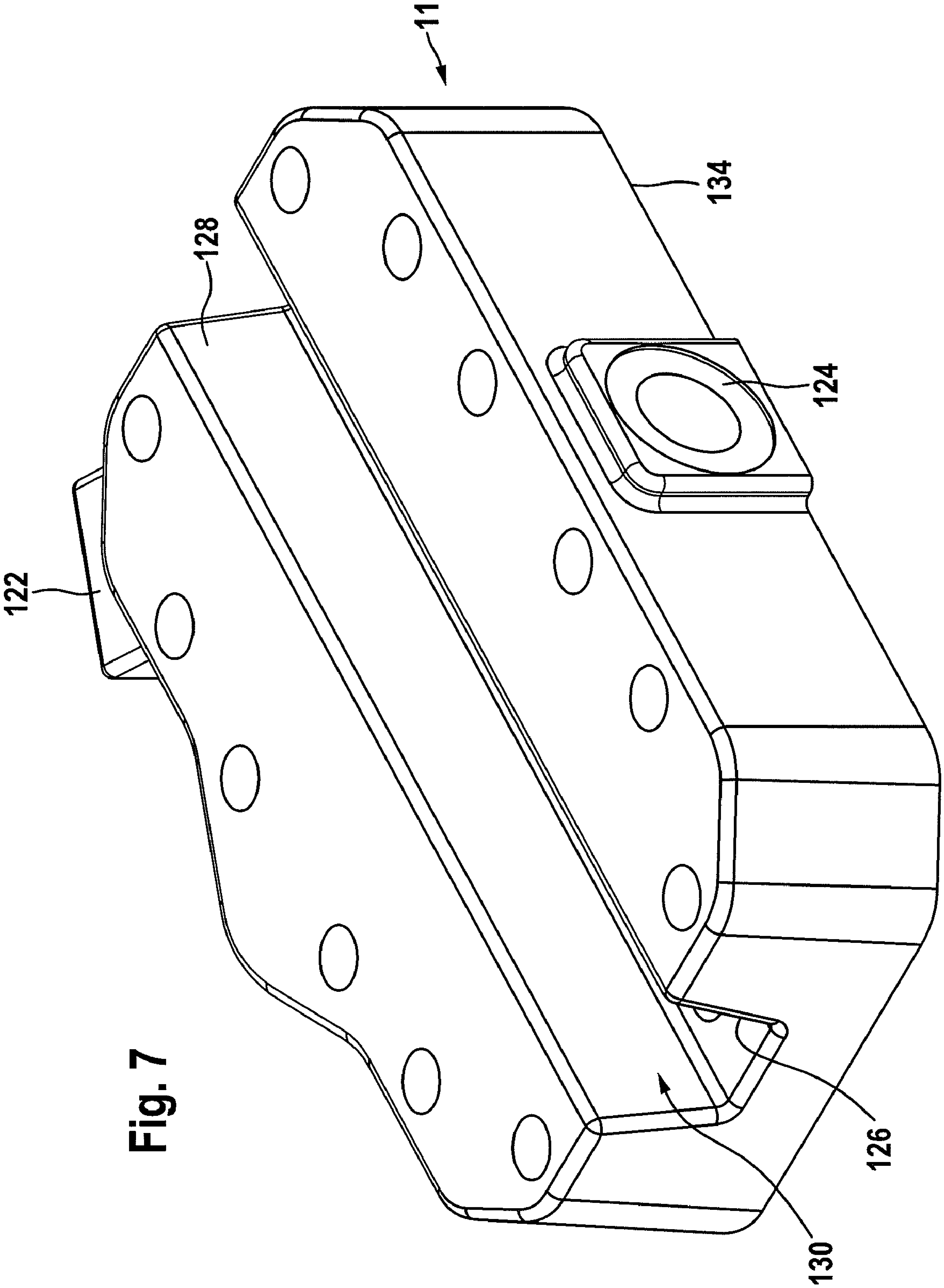


Fig. 7

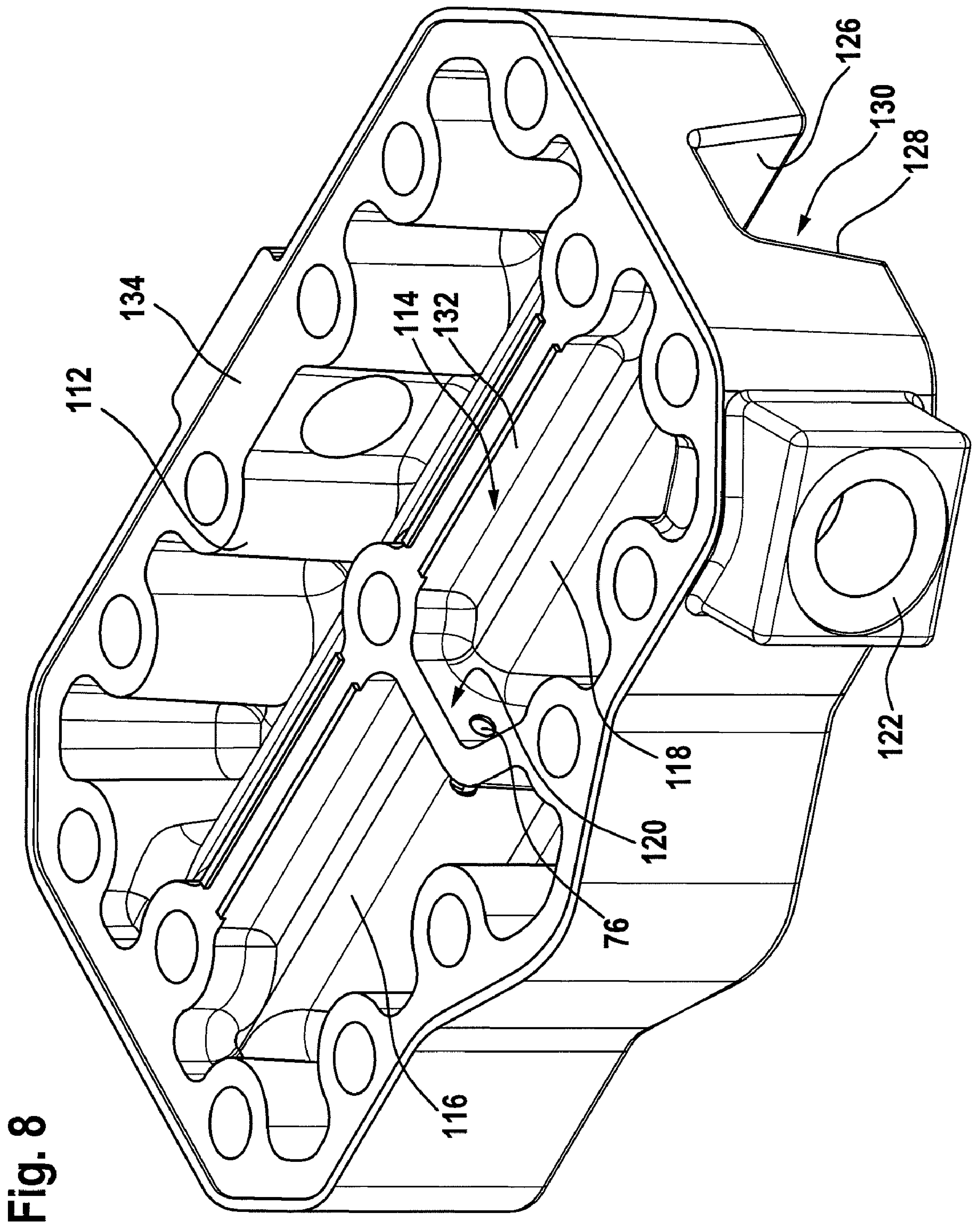


Fig. 8

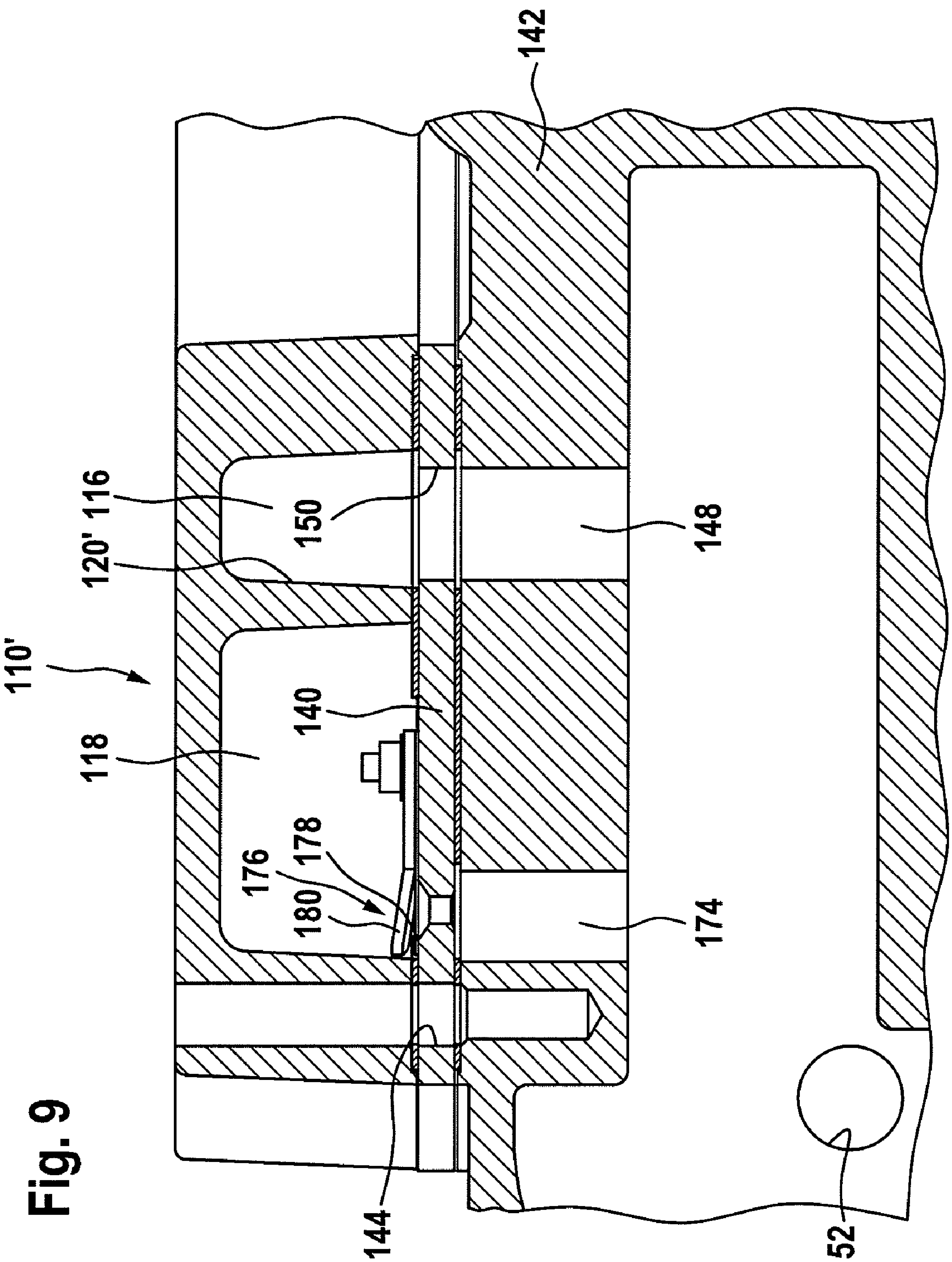
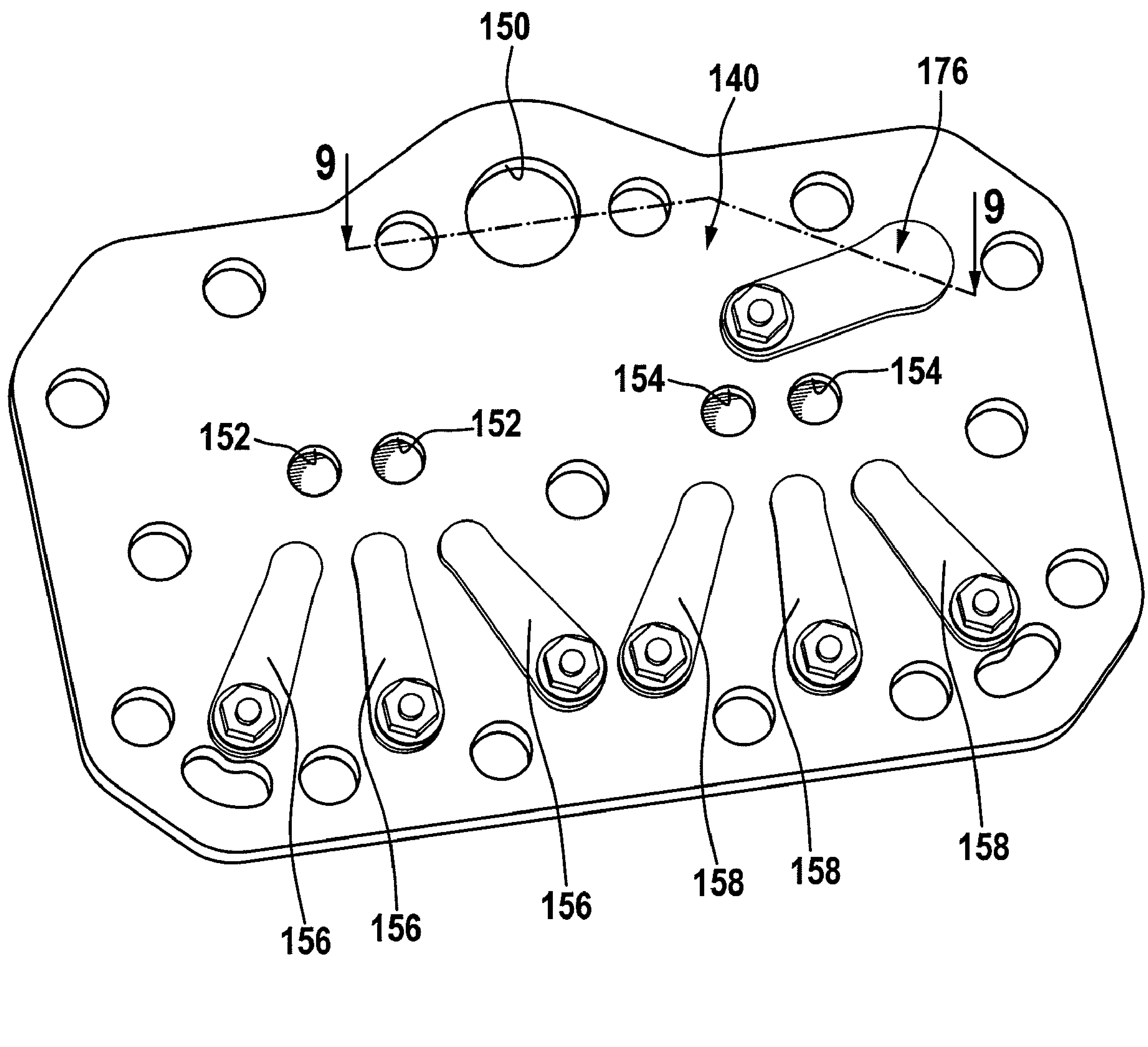


Fig. 9

Fig. 10



REFRIGERATION SYSTEM

CROSS-REFERENCE TO RELATED PATENT APPLICATIONS

This application is a continuation of international application No. PCT/EP2006/000581 filed on Jan. 24, 2006 and claims the priority and benefit of German application No. 10 2005 009 173.3 filed on Feb. 17, 2005, the teachings and disclosure of which are hereby incorporated in their entirety by reference thereto.

BACKGROUND OF THE INVENTION

The invention relates to a refrigeration system comprising a refrigerant circuit, in which a main mass flow of a refrigerant—preferably carbon dioxide—is guided, a heat exchanger arranged in the refrigerant circuit on the high pressure side, an expansion cooling device which is arranged in the refrigerant circuit, cools the main mass flow of the refrigerant in the active state and thereby generates an additional mass flow of gaseous refrigerant, a reservoir for liquefied refrigerant arranged in the refrigerant circuit, at least one expansion unit for liquefied refrigerant of the main mass flow, this expansion unit being arranged in the refrigerant circuit and having an expansion element and a post-connected heat exchanger on the low pressure side which makes refrigerating capacity available, i.e., increases the enthalpy of the refrigerant, and at least one refrigerant compressor which is arranged in the refrigerant circuit and has a main compressor stage and at least one additional compressor stage driven together with the main compressor stage, these two stages compressing refrigerant to high pressure, wherein the main compressor stage and the at least one additional compressor stage can be used such that either the main compressor stage compresses refrigerant from the main mass flow and the additional compressor stage refrigerant from the additional mass flow or the main compressor stage and the additional compressor stage compress refrigerant from the main mass flow.

Refrigeration systems of this type are known from the state of the art, wherein they are designed for customary refrigerants.

Refrigeration systems of this type are described, for example, in EP 0 180 904 A2.

Proceeding from this known state of the art, the object underlying the invention is to create a refrigeration system which may be adapted to different operating conditions in an optimum manner.

SUMMARY OF THE INVENTION

This object is accomplished in accordance with the invention, in a refrigeration system of the type described at the outset, in that at least two refrigerant compressors are arranged in the refrigerant circuit and can be switched on individually for the purpose of compressing the main mass flow, that at least two of the refrigerant compressors each have at least one additional compressor stage, that each of the additional compressor stages can be used optionally for the compression of refrigerant from the main mass flow or for the compression of refrigerant from the additional mass flow and that a control is provided, with which such a number of additional compressor stages for the compression of refrigerant from the additional mass flow can be switched on in a first operating mode as a function of operating conditions that the expansion cooling device liquefies the main mass flow and reduces its enthalpy.

The advantage of the solution according to the invention is to be seen in the fact that this creates the possibility, on account of the variable switchability of the additional compressor stages, of adapting the liquefying of the main mass flow and the reduction in enthalpy to the various operating conditions and, therefore, of always keeping them in an optimum range.

It is particularly favorable when the expansion cooling device reduces the enthalpy of the main mass flow by at least 10%.

It is even more advantageous when the expansion cooling device reduces the enthalpy of the main mass flow by at least 20%. The refrigeration system can be used particularly favorably when the first operating mode corresponds to a supercritical operation, for example, with carbon dioxide as refrigerant.

A supercritical operation is to be understood such that the refrigerant compressed to high pressure cannot be cooled in the heat exchanger on the high pressure side to a temperature which corresponds to an isotherm passing through the boiling point curve and saturation curve of the refrigerant but rather can merely be cooled to a temperature which corresponds to an isotherm extending outside the boiling point curve and saturation curve and so the refrigerant is not liquefied.

Furthermore, a particularly favorable embodiment provides for the expansion cooling device to convert the main mass flow into a thermodynamic state, the pressure and enthalpy of which are lower than pressure and enthalpy of a maximum of the saturation curve or boiling point curve in an enthalpy/pressure diagram.

The thermodynamic state of the main mass flow brought about by the expansion cooling device is preferably close to the boiling point curve of the enthalpy/pressure diagram, in particular, essentially on the boiling point curve or at an enthalpy which is lower than the enthalpy corresponding to the boiling point curve at the respective pressure.

The expansion cooling device may be designed, in principle, in any optional manner.

A particularly favorable solution provides, however, for the expansion cooling device to have an expansion valve for the expansion of refrigerant to an intermediate pressure and for the intermediate pressure of the expansion cooling device to be adjustable by switching on the suitable number of additional compressor stages.

Furthermore, the expansion cooling device could operate, for example, such that only an expansion of the refrigerant forming the additional mass flow takes place.

It is, however, even more advantageous when the expansion valve of the expansion cooling device expands refrigerant of the main mass flow and of the additional mass flow to the intermediate pressure.

With respect to the arrangement of the reservoir for the liquid refrigerant, no further details have so far been given. One particularly favorable solution provides, for example, for the expansion cooling device to also comprise the reservoir for the liquid refrigerant of the main mass flow and, therefore, the construction of the refrigeration system according to the invention is simplified.

A solution which is particularly preferred from a constructional point of view provides for the expansion valve to transfer the expanded refrigerant from the main mass flow and the additional mass flow into a container, in which the reservoir for the liquid refrigerant of the main mass flow is formed, over which a vapor chamber is located, from which the refrigerant forming the additional mass flow is then discharged so that part of the refrigerant vaporizes and, as a result, cools or even supercools the main mass flow.

An additional, advantageous embodiment of the refrigeration system according to the invention provides for the expansion cooling device to be in the inactive state in a second operating mode and to bring about no cooling of the main mass flow.

This means that, in this case, no additional mass flow of refrigerant results and, therefore, the refrigeration system according to the invention can be operated in the conventional, known manner by way of a circuit of the entire refrigerant in the form of the main mass flow.

It is expediently provided in such a second operating mode of the refrigeration system for all the additional compressor stages to compress refrigerant of the main mass flow.

Furthermore, it is provided in a second operating mode of the refrigeration system according to the invention for the reservoir for liquid refrigerant of the main mass flow to be subject to high pressure.

It is provided, in particular, in one embodiment for the second operating mode to correspond to a subcritical operation of the refrigeration system.

A subcritical operation of the refrigeration system within the meaning of the solution according to the invention is to be understood such that such a strong cooling of the refrigerant compressed to high pressure is possible in the heat exchanger on the high pressure side that this refrigerant is converted into a thermodynamic state which is below the saturation curve or boiling point curve, i.e., in the range of the coexistence of liquid and vapor and is, therefore, cooled in such a manner that the refrigerant is liquefied by the heat exchanger on the high pressure side.

In order to be able to always operate the refrigeration system according to the invention at optimum conditions, in particular, in adapting to the refrigerating capacity required, it is provided for the control to control the refrigerant compressors in accordance with the refrigerating capacity required, i.e., the refrigerant compressors can either be operated with a variable rotational speed and/or can be switched on or off.

In this respect, it is particularly expedient when the control is in a position to switch the refrigerant compressors on or off individually in accordance with the refrigerating capacity required, i.e., for it to be possible, by switching the at least two refrigerant compressors in the refrigerant circuit on or off individually, to adapt the compressor capacity to the refrigerating capacity required and, therefore, always operate the refrigeration system according to the invention in an optimum manner.

With respect to the configuration of the additional compressor stages in relation to the respective main compressor stage, no further details have so far been given. It is, for example, particularly favorable when each refrigerant compressor with additional compressor stage is dimensioned such that the mass flow of refrigerant of the additional mass flow compressed by the additional compressor stage corresponds at the most to the mass flow of refrigerant of the main mass flow compressed by the main compressor stage in this refrigerant compressor.

Furthermore, the possibilities provided by the control for adjusting the additional mass flow and the intermediate pressure may be utilized advantageously in that the refrigerant compressors with additional compressor stage are dimensioned such that the additional compressor stages of different refrigerant compressors compress different mass flows of refrigerant of the additional mass flow.

As a result, a considerable variation of the additional mass flow to be compressed can be brought about by way of a suitable selection of the additional compressor stages provided for the compression of refrigerant of the additional mass flow, in particular, by way of a suitable combination of additional compressor stages configured for different mass flows without the capacity of the main compressor stages needing to be altered for this purpose.

Since, in the case of refrigeration systems which are intended to operate in the supercritical range, a very great difference in pressure must be generated during the compression of the refrigerant, it is preferably provided for the refrigerant compressors with additional compressor stage to be reciprocating compressors.

In the case of reciprocating compressors of this type, each of the refrigerant compressors with additional compressor stage is expediently designed such that this has at least one cylinder for the additional compressor stage and at least one cylinder for the main compressor stage.

A refrigeration system of this type may be realized particularly favorably when the number of cylinders for the main compressor stage is greater than the number of cylinders for the additional compressor stage in each refrigerant compressor with additional compressor stage.

Furthermore, a solution of the refrigeration system according to the invention which is particularly favorable with respect to the variable adjustability of the additional mass flow provides, in the case of the refrigerant compressors with additional compressor stage, for the additional compressor stages of different refrigerant compressors to have a different volumetric displacement so that, as a result, a particularly broad range of volumetric displacements for the additional mass flow is also available for selection in different combinations of the additional compressor stages.

Furthermore, an additional solution which is suitable with respect to its variability provides for the ratio of the volumetric displacement of the additional compressor stage to the volumetric displacement of the main compressor stage for each refrigerant compressor with additional compressor stage to be different in relation to at least one of the other refrigerant compressors with additional compressor stage so that not only the volumetric displacements of the additional compressor stages may be combined by suitable selection and combination with one another to form as great a range of variation as possible but also the volumetric displacements of the main compressor stages.

A further, advantageous embodiment of the refrigeration system according to the invention provides for the reservoir for liquefied refrigerant to operate at an intermediate pressure in the first operating mode and for an additional expansion unit with an expansion element and a post-connected heat exchanger making refrigerating capacity available to be provided between the heat exchanger on the high pressure side which cools the refrigerant and the reservoir for liquefied refrigerant. The degree of thermodynamic effectiveness of the refrigeration system according to the invention may be improved even further with this additional expansion unit since the vaporization temperature in this additional expansion unit is higher which presupposes that the refrigerating capacity available can be used at a higher temperature level, for example, for air cooling or air conditioning.

A considerably improved degree of thermodynamic effectiveness can be achieved at supercritical operating conditions, in particular, in all the preceding embodiments. Moreover, the refrigerating capacity for a defined compressor volumetric displacement is greater and the characteristic capacity curve is flatter in relation to the surrounding temperature which has a positive effect on the regulating characteristics of the refrigeration system.

The reason for the greater cost efficiency during supercritical operation is, in particular, the fact that the vaporization of the additional mass flow is brought about at a higher level of pressure than the vaporization in the heat exchangers of the expansion units on the suction side. This leads to an improvement in the degree of thermodynamic effectiveness resulting in reduced energy requirements for a defined refrigerating capacity.

Cooling of the refrigerant of the main mass flow at saturation pressure up to the boiling point curve or saturation curve is brought about, in particular, due to the expansion of the main mass flow and of the additional mass flow in conjunction with the additional mass flow being drawn off by suction. As a result, the difference in enthalpy for the vaporization and overheating is increased. The percentage increase in the difference in enthalpy is higher than the proportion of compressor capacity which must be used for the compression of the additional mass flow. Apart from the improvement in the degree of effectiveness mentioned before, this also leads to a greater refrigerating capacity—in relation to an identical total volumetric displacement of the refrigeration system.

Furthermore, it is of advantage in the case of the refrigeration system according to the invention, in particular, when carbon dioxide is used as refrigerant when the refrigerant compressor has cylinder heads, with which inlet chambers and outlet chambers are essentially thermally decoupled so that essentially no heating up of the inlet chambers with the refrigerant to be drawn in by suction takes place as a result of the heating of the refrigerant during the compression to high pressure and the heating up of the outlet chambers connected therewith and, therefore, there is no negative influence on the compressor capacity.

In order to be able to use the additional compressor stages either for the compression of the main mass flow or for the compression of the additional mass flow it is, in principle, conceivable to provide controlled valves which supply the additional compressor stages either with refrigerant from the main mass flow or from the additional mass flow.

A solution which is particularly simple from a constructional point of view provides, however, for a check valve to be provided for connecting an inlet chamber of the additional compressor stage to the low pressure connection of the main compressor stage so that the additional compressor stage compresses refrigerant of the main mass flow automatically when the additional mass flow is interrupted.

A particularly simple solution provides in this respect for the check valve to connect the inlet chamber of the additional compressor stage to the inlet chamber of the main compressor stage.

Another advantageous solution provides for the check valve to be provided in a valve plate of the respective refrigerant compressor. This solution has the advantage that the valve plate which is already equipped with valves need merely be provided with an additional check valve and, therefore, the check valve is particularly easy to mount.

In this respect, it is particularly expedient when a connecting channel between the low pressure connection and the check valve runs in a cylinder housing and can be integrally formed in it in the same way as the inlet channel for supplying the main compressor stage with refrigerant supplied via the low pressure connection.

Additional features and advantages of the invention are the subject matter of the following description as well as the drawings illustrating several embodiments.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** shows a schematic illustration of a first embodiment of a refrigeration system according to the invention;

FIG. **2** shows a schematic illustration of one of the refrigerant compressors used in the refrigeration system according to the invention in accordance with the first embodiment and comprising main compressor stage and additional compressor stage;

FIG. **3** shows an illustration of the pressure [P] over the enthalpy [h] in the case of a subcritical cyclic process which can be realized with the first embodiment and a possible supercritical cyclic process not, however, corresponding to the invention;

FIG. **4** shows an illustration of the pressure [P] over the enthalpy [h] in the case of a cyclic process according to the invention which can be carried out with the first embodiment of the solution according to the invention in the supercritical range with expansion of the refrigerant compressed to high pressure to an intermediate pressure and simultaneous reduction of the enthalpy due to an additional mass flow being drawn off by suction;

FIG. **5** shows a schematic illustration of a refrigerant compressor in a second embodiment of the refrigeration system according to the invention;

FIG. **6** shows a schematic illustration of a third embodiment of a refrigeration system according to the invention;

FIG. **7** shows a perspective illustration of a cylinder head of a first, preferred embodiment of a refrigerant compressor for a refrigeration system according to the invention;

FIG. **8** shows a perspective view of the cylinder head according to FIG. **7** with an underside thereof pointing upwards;

FIG. **9** shows a partial section through a second, preferred embodiment of a refrigerant compressor for the refrigeration system according to the invention; and

FIG. **10** shows a perspective illustration of a valve plate of the second, preferred embodiment of the refrigerant compressor according to FIG. **9**.

DETAILED DESCRIPTION OF THE INVENTION

One embodiment of a refrigeration system illustrated in FIG. **1** comprises a refrigerant circuit which is designated as a whole as **10** and in which several, for example, three refrigerant compressors **12***a* to **12***c* are arranged, the high pressure connections **14***a* to **14***c* of which are connected to a high pressure line **16** of the refrigerant circuit **10**.

The high pressure line **16** leads to a heat exchanger **18** on the high pressure side which cools the refrigerant compressed to high pressure PH, for example, with a stream **20** of cooling agent, wherein the cooling agent is preferably ambient air which flows through the heat exchanger **18**.

It is, however, also conceivable to provide another cooling agent, for example, water or the like instead of ambient air.

An additional high pressure line **22** leads from the heat exchanger **18** to an expansion valve **24** and to a bypass valve **26** which is connected in parallel to the expansion valve **24**, both of which open into a container **28** which is designed such that it comprises a reservoir **30** for liquid refrigerant, in which a volume **32** of liquid refrigerant is always present which—as will be described in detail in the following—represents a buffer volume for liquid refrigerant in the refrigerant circuit **10**.

A line **34** leads from the reservoir **30** to expansion units **40**, for example, four expansion units **40***a* to **40***d* which are connected in parallel.

The line **34** is connected to the reservoir **30** in such a manner that it conveys essentially only liquid refrigerant to the expansion units **40** and they can, therefore, be operated and, in particular, regulated in the known manner since an expansion of liquid refrigerant, essentially without any proportion of gas, always takes place.

The regulation of expansion units **40** which are supplied with liquid refrigerant corresponds to the type of regulation in the case of known refrigeration systems.

Each of the expansion units **40** comprises a stop valve **42**, an expansion valve **44** which expands the liquid refrigerant and a heat exchanger **46** on the low pressure side which is in a position, on account of the expanded refrigerant, to provide refrigerating capacity, as designated by the arrow **48**.

The heat exchangers **46** of the expansion units **40** connected in parallel are connected to a common low pressure line **50** which leads to low pressure connections **52***a* to **52***c* of the refrigerant compressors **12***a* to **12***c*.

The sum of all the branch mass flows **54***a*, **54***b*, **54***c* and **54***d* of the refrigerant, which pass through the expansion units **40** and are collected by the low pressure line **50**, form a main mass flow **56** of the refrigerant circuit **10** which is again divided up, for its part, into branch mass flows **58***a*, **58***b* and **58***c* which are drawn in by the refrigerant compressors **12***a* to **12***c* via the lower pressure connections **52***a* to **52***c* and compressed to high pressure PH in order to exit again through the high pressure connections **14***a* to **14***c* of the refrigerant compressors **12**.

Since the branch mass flows **54***a* to **54***d* have been removed from the line **34**, the main mass flow **56** also flows through the line **34** following the reservoir **30** and is then allotted again to the branch mass flows **54***a* to **54***d*.

As illustrated in FIG. **2**, each of the refrigerant compressors **12** is designed, for example, as a reciprocating compressor and comprises a cylinder housing **60**, in which altogether four cylinders **62***a* to **62***d* are, for example, provided, in which refrigerant can be compressed by means of pistons **64***a* to **64***d* moved oscillatingly.

In a refrigerant compressor **12** designed in such a manner in accordance with the invention, not all the cylinders **62***a* to **62***d* operate as a uniform compressor stage but rather the cylinders **62***a* to **62***c* are, for example, combined to form a main compressor stage **66**, in which these three cylinders **62***a* to **62***c* operate in parallel, i.e., all three cylinders **62***a* to **62***c* draw in refrigerant via the respective low pressure connection **52** and deliver refrigerant compressed to high pressure PH to the respective high pressure connection **14**.

Furthermore, the cylinder **62***d*, which is driven by a drive motor **68** together with the remaining cylinders of the main compressor stage **66** and in the same way as them, is operated as a separate additional compressor stage **70** which is likewise connected to the high pressure connection **14** on the output side but is in a position to draw in refrigerant either via an additional connection **72** or via the low pressure connection **52**.

In the simplest case, a check valve **76** is provided in the connecting channel **74** running between the additional connection **72** and the low pressure connection **52** and this blocks the connecting channel **74** when the pressure at the additional connection **72** is higher than that at the low pressure connection **52** and so the connecting channel **74** is blocked when refrigerant is present at the additional connection **72** at a higher pressure than at the lower pressure connection **52** and, therefore, the additional compressor stage **70** draws in refrigerant via the additional connection **72**. A controlled valve can, however, also be provided.

If, however, the additional connection **72** is closed or blocked so that no refrigerant can be drawn in via this connection, the check valve **76** opens and the additional compressor stage **70** draws in refrigerant via the lower pressure connection **52** and compresses this to high pressure PH in the same way as the main compressor stage **66**.

As illustrated in FIG. **1**, the additional connections **72***a* to **72***c* of the refrigerant compressors **12***a* to **12***c* are each connected via stop valves **80***a* to **80***c* to a distributor line **82** which opens into the container **28**, namely such that it is in a position to discharge vaporized refrigerant out of a vapor chamber **84** of the container **28**.

The vaporized refrigerant discharged by the distributor line **82** from the container **28** forms an additional mass flow **86** which can be distributed by the distributor line **82** to the additional compressor stages **70** in order to be compressed by them to high pressure PH.

The additional mass flow **86** can therefore be controlled due to the fact that individual ones of the stop valves **80***a* to **80***c* are opened or closed.

For the purpose of controlling the refrigeration system, a control designated as **90** is provided altogether and this is in a position to activate the individual stop valves **80***a* to **80***c* individually.

If all the stop valves **80***a* to **80***c* are closed, no additional mass flow **86** flows through the distributor line **82** and no compression of an additional mass flow **86** takes place in the additional compressor stages **70** and so only the main mass flow **56** is compressed and expanded altogether in the refrigerant circuit **10** with all the cylinders **62**.

If, however, one of the stop valves **80***a* to **80***c* is open or all the stop valves **80***a* to **80***c* are open, the additional mass flow **86** flows through the distributor line **82**, is supplied to the additional compressor stages **70** which are connected to the distributor line **82** via the opened stop valves **80***a* to **80***c* and is, therefore, compressed by the corresponding additional compressor stages **70** of the respective refrigerant compressors **12** such that the additional mass flow **86** flows not only through the high pressure line **16** but also through the heat exchanger **18** on the high pressure side in addition to the main mass flow **56** and is supplied to the container **28** via the additional high pressure line **22**, wherein a separation takes place between the main mass flow **56** and the additional mass flow **86** in the container **28** to the effect that the main mass flow **56** is supplied to the expansion units **40** via the line **34** whereas the additional mass flow **86** is supplied to the corresponding additional compressor stages **70** via the distributor line **82** and does not, therefore, flow through the expansion units **40**.

A refrigeration system designed in this way may be operated as follows with, in particular, carbon dioxide (CO_2) used as refrigerant:

If an adequately vigorous cooling of the refrigerant, which has been compressed to high pressure PH, in the heat exchanger **18** on the high pressure side is possible, the refrigeration system may be operated in the so-called subcritical cyclic process. With carbon dioxide as refrigerant, this presupposes that the temperature of the cooling agent **20** supplied to the heat exchanger **18** on the high pressure side is in the order of magnitude of approximately 23° C. or below. In this case, the cooling of the refrigerant compressed to high pressure PH leads to a liquefying thereof and so the bypass valve **26** is opened by the control **90** and the liquid refrigerant is supplied directly to the reservoir **30** for liquid refrigerant from the additional high pressure line **22**.

This liquid refrigerant then forms the main mass flow **56** which is distributed to the individual expansion units **40** via the line **34** insofar as they have been switched on by the control **90**, i.e., the stop valves **42***a* to **42***d* are open.

The activation of the individual expansion units **40***a* to **40***d* is brought about irrespective of whether or not refrigerating capacity **48** is intended to be made available in the area of the respective heat exchanger **46** on the low pressure side.

The refrigerant expanded in the individual expansion units **40***a* to **40***d* is then supplied to the individual low pressure connections **52***a* to **52***c* of the individual refrigerant compressors **12***a* to **12***c* via the low pressure line **50**.

The control **90** does not necessarily operate all the refrigerant compressors **12***a* to **12***c* in the full load range but rather can operate either individual ones of the refrigerant compressors **12***a* to **12***c* in the full load range or individual ones or all of the refrigerant compressors **12***a* to **12***c* in the partial load range, i.e., with a reduced rotational speed of the respective drive motor **68**. It is, however, also possible to switch off individual ones of the refrigerant compressors **12***a* to **12***c* completely on the part of the control **90**, for example, when only some of the expansion units **40***a* to **40***d* are intended to have refrigerating capacity made available at their respective heat exchanger **46**.

Moreover, the control closes the stop valves **80***a* to **80***c* in the subcritical range so that the additional compressor stages **70** of all the refrigerant compressors **12***a* to **12***c* draw in refrigerant from the main mass flow **56** via the respective check valve **76** and compress it to high pressure PH.

Such a cyclic process for the subcritical operation is illustrated in FIG. **3** by the dashed lines, wherein the state at point A represents the beginning of compression of refrigerant from the main mass flow **56** by the respective refrigerant compressor **12** which is terminated at the state at point B.

Proceeding from the state at point B, the refrigerant compressed at high pressure PH is cooled as far as a state at point C which is approximately on the saturation curve or boiling point curve **96** for carbon dioxide as refrigerant.

This refrigerant which is now cooled but liquefied in the heat exchanger **18** can now be supplied in this state to the individual expansion units **40**, wherein an isenthalpic expansion of the refrigerant takes place as a result of the expansion valve **44** of each of the expansion units **40** which leads to a reduction in the pressure combined with a reduction in the temperature and so the state at point D in FIG. **3** is reached.

Proceeding from the state at point D, the refrigerating capacity **48** can now be made available in the respective heat exchanger **46** on the low pressure side as a result of an increase in enthalpy until the state at point A is again reached which, with respect to enthalpy and pressure, represents the refrigerant which is supplied to the low pressure connections **52** of the refrigerant compressors **12** via the low pressure line **50**.

If no cooling agent is, however, available which is a position to cool the refrigerant to a temperature in the order of magnitude of 23° C. but rather the cooling agent available is one which merely allows cooling to higher temperatures of the refrigerant, for example, over 31°, only a so-called supercritical cyclic process (illustrated in FIG. **3** by solid lines) would be possible with the refrigeration system according to FIG. **1**, with an open bypass valve **26** and inoperative expansion valve **24**; with this process the refrigerant would have had to be compressed to a higher pressure in accordance with the state at point B' in FIG. **3**, wherein a subsequent cooling in the heat exchanger **18** on the high pressure side leads to a state at point C' in FIG. **3** which is outside the saturation curve **96**.

The refrigerant in the state at point C' in FIG. **3** is still gaseous. A subsequent, isenthalpic expansion of the refrigerant in the individual expansion units **40** would then lead to the state at point D' in FIG. **3**, wherein the consequence of this would be the fact that gaseous refrigerant would be supplied to the expansion valves **44** of the expansion units **40** and gaseous refrigerant would have to be expanded. Such an expansion of a gaseous refrigerant is subject to different regulating characteristics and so, as a result, the regulating characteristics known so far for the expansion valves **44** are not suitable.

For this reason, no supercritical cyclic process from point A to B' to C' and D' and then again to A, as illustrated in FIG. **3**, takes place during supercritical operation of the refrigeration system but rather the bypass valve **26** is closed by the control **90** and the expansion valve **24** is activated so that the refrigerant entering the container **28** from the additional high pressure line **22** can be expanded by the expansion valve **24** to an intermediate pressure PZ in accordance with the state Z in FIG. **4**. In this respect, the temperature may be reduced, in addition, in the case of the intermediate pressure PZ due to vaporization of refrigerant to such an extent, and, therefore, the enthalpy also be reduced, that liquid refrigerant of the main flow **56**, the state of which corresponds to the state at point C on the boiling point curve **96** in FIG. **4**, is present in the container **28**.

In order to be able to arrive at the state at point C from the state at point C' in FIG. **4**, it is necessary to predetermine the intermediate pressure PZ in the container **28** and to stabilize this intermediate pressure PZ by drawing off the vaporized refrigerant, wherein the vaporized refrigerant results in the additional mass flow **86** which must be discharged from the vapor chamber **84** of the container in order to be able to keep the intermediate pressure PZ at the desired level.

The state at point C is at a value of the enthalpy [h] which is lower by more than 20% in relation to a maximum **98** of the boiling point curve **96** and which is reached due to the vaporization of the refrigerant forming the additional mass flow, wherein the state at point C in FIG. **4** is either essentially on the boiling point curve **96** or, where applicable, in the case of additional cooling, e.g., via a heat exchanger which has expanded main mass flow passing through it at a somewhat lower enthalpy than the enthalpy of the state at point C.

For this purpose, the control **90** must open at least some of the stop valves **80***a* to **80***c* or all the stop valves **80***a* to **80***c* in order, as a result, to cause refrigerant to be drawn in from the additional flow **86** by the additional compressor stages **70** in order to maintain the intermediate pressure PZ in the container **28** and to be compressed to high pressure PH.

As a result, the refrigerant of the main mass flow **56** may be supplied to the individual expansion units **42***a* to **42***c* via the line **34** and converted, due to isenthalpic expansion in the expansion units **40** by means of the expansion valves **44**, into the state designated in FIG. **4** as point D, in which the output of refrigerating capacity **48** is possible with an increase in enthalpy up to the state at point A in the respective heat exchanger **46** on the low pressure side, wherein it is apparent in a comparison with FIG. **3** that the refrigerating capacity available is greater than in a supercritical cyclic process in accordance with the states at points A, B', C', D' in FIG. **3**.

The advantage of the idea according to the invention is to be seen in the fact that it is possible to select the high pressure PH in an optimum manner in accordance with the course of the isotherms of the refrigerant used without the subsequent expansions needing to be taken into consideration.

Furthermore, the intermediate pressure PZ may likewise be optimized by way of suitable variation of the additional mass flow, namely such that the percentage reduction in the enthalpy of the main mass flow is higher than the proportion of displacement capacity which is required for the additional mass flow in the total displacement capacity of the compressor and so the losses with respect to displacement capacity caused by compression of the additional mass flow are overcompensated by the reduction in the enthalpy of the main mass flow.

The cyclic process carried out for maintaining the intermediate pressure PZ as a result of compression of the additional mass flow **86** is illustrated in FIG. **4** by dashed lines and extends from the state at point Z as a result of an increase in enthalpy of the vaporized refrigerant to the state at point A" and from the state at point A" to the state at point B" which is, again, at the high pressure PH and from the state at point B" to the state at point C' and from the state at point C' to the state at point Z.

In the case of such a supercritical cyclic process, as illustrated in FIG. **4**, between the states at the points A, B', C', C and D, the additional mass flow **86** which occurs is not constant in relation to the main mass flow **56** when the intermediate pressure PZ is intended to be adjusted in an optimized manner but varies depending on how many expansion units **40** are activated in the refrigerant circuit **10** and depending on how high the temperature of the cooling agent **20** is which is supplied to the heat exchanger **18** on the high pressure side.

In order to have an intermediate pressure PZ corresponding to optimized operating conditions with the most varied of operating conditions and, consequently, to be able to compress an additional mass flow **86** maintaining this intermediate pressure PZ via the additional compressor stages **70**, the additional compressor stages **70** of the refrigerant compressors **12** are designed in such a manner that an optimized supercritical operation is still possible with a maximum output of refrigerating capacity by all the expansion units **40** and with a maximum temperature of the cooling agent **20** and the additional mass flow **86** thereby resulting can be compressed to high pressure PH by the entirety of the active additional compressors stages **70** to maintain a suitable level of the intermediate pressure PZ.

If more favorable operating conditions are present, proceeding from this operating state, the control **90** can either reduce the rotational speed of the drive motors **68** of one or more of the refrigerant compressors **12** or switch off one of the refrigerant compressors **12**, wherein, as a result, not only the compressor capacity of the main compressor stage of this refrigerant compressor **12** is dispensed with but also the compressor capacity of the additional compressor stage **70**.

If, however, the operating conditions change to the extent that, for example, cooling agent **20** with a lower temperature is available, the additional mass flow **86** is altered since less refrigerant has to be vaporized in order to obtain liquid refrigerant in the state at point C according to FIG. **4** at a suitable intermediate pressure PZ.

In this case, the control **90** has the possibility of adapting the compressor capacity of the additional compressor stages **70** to the smaller additional mass flow **86** required by closing one or two of the stop valves **80**_a_ to **80**_c_ and, therefore, of maintaining an optimized intermediate pressure PZ in the container **28**.

The additional compressor stages **70**, the stop valves **80** of which have been closed, then draw in refrigerant of the corresponding low pressure connection **52** and, therefore, compress refrigerant from the respective main mass flow **56**.

The idea according to the invention therefore allows an optimum adaptation of the intermediate pressure PZ by adapting the compressor capacity of the additional compressor stages **70**_a_ to **70**_c_ required for the compression of the additional mass flow **86** independently of the compressor capacity of the main compressor stages **66**.

In principle, it would be possible to design the refrigerant compressors **12**_a_ to **12**_c_ in an identical manner so that each main compressor stage **66** and each additional compressor stage **70** can generate the same compressor capacity.

It is, however, even more advantageous for the adaptation to different operating conditions when the refrigerant compressors **12**_a_ to **12**_c_ are designed such that, for example, a second one of the refrigerant compressors **12**_a_ to **12**_c_ has double the compressor capacity of the first refrigerant compressor and a third refrigerant compressor double the compressor capacity of the second refrigerant compressor, wherein the doubling of the compressor capacity relates not only to the main compressor stages **66** but also the additional compressor stages **70**.

As a result, it is possible to obtain different multiples of the compressor capacity of the first refrigerant compressor as a result of different combinations of the first, second and third refrigerant compressors, for example, double the compressor capacity of the first refrigerant compressor solely by operating the second refrigerant compressor, treble the compressor capacity of the first refrigerant compressor by operating the first and second refrigerant compressors, four times the refrigerating capacity by operating the third refrigerant compressor and five times the refrigerating capacity by operating the third refrigerant compressor in combination with the first refrigerant compressor as well as seven times the compressor capacity by a combination of the first, second and third refrigerant compressors.

With respect to the compressor capacity of the additional compressor stages **70**, additional possibilities for variation are also conceivable, namely to the extent that, for example, the maximum capacity of the additional compressor stages **70** for the additional mass flow **86** is available and this corresponds to seven times the compressor capacity of the first refrigerant compressor when all three refrigerant compressors **12**_a_ to **12**_c_ are operated.

It is, however, also conceivable to use optional, whole number multiples of the compressor capacity of the additional compressor stage **70** of the first refrigerant compressor with analogous use of the procedure described above by opening stop valves **80** and connecting the individual additional compressor stages **70** of the individual refrigerant compressors **12** to the distributor line **82** for the purpose of compressing the additional mass flow **86**.

In a second embodiment of a refrigeration system according to the invention, illustrated in FIG. **5**, the refrigerant compressors **12'** are designed, for example, such that they have two additional compressor stages $\mathbf{70}_1$ and $\mathbf{70}_2$ which each have their own additional connections $\mathbf{72}_1$ and $\mathbf{72}_2$.

For example, the cylinder **62**_c_ forms the additional compressor stage $\mathbf{70}_2$ and the cylinder **62**_d_ the additional compressor stage $\mathbf{70}_1$ while the cylinders **62**_a_ and **62**_b_ form the main compressor stage **66**.

Such a construction of one of the refrigerant compressors **12** or all the refrigerant compressors **12** creates an even greater variability with respect to the compressor capacity available for the compression of the additional mass flow **86** since the individual additional compressor stages $\mathbf{70}_1$ and $\mathbf{70}_2$ can be selectively connected to the distributor line **82** individually or together by opening the corresponding stop valves **80** or can be used for the purpose of compressing refrigerant of the main mass flow **56**.

As for the rest, the second embodiment of the refrigeration system according to the invention corresponds to the first embodiment and so reference can be made in full to the description of the first embodiment of the refrigeration system according to the invention.

A third embodiment of the refrigeration system according to the invention, illustrated in FIG. **6**, is also based on the first embodiment of the refrigeration system according to the invention, wherein the same parts are given the same reference numerals and so with respect to the description thereof reference is made in full to the comments on the first embodiment.

In the third embodiment, an additional expansion unit **100** is connected in parallel to the bypass valve **26** and the expansion valve **24**.

The additional expansion unit **100** comprises, for its part, a stop valve **102**, an expansion valve **104** and a heat exchanger **106** on the high pressure side, from which refrigerating capacity can be discharged, as designated by an arrow **108**.

It is likewise possible with this additional expansion unit to expand refrigerant from the high pressure line **22** and, therefore, to obtain refrigerating capacity **108** which is available externally, wherein the refrigerant is merely expanded to the intermediate pressure PZ present in the container **28**.

It is, therefore, possible to operate a heat exchanger **106** working at a higher temperature level, in addition, during supercritical operation and, as a result, increase the degree of efficiency of the refrigeration system.

The refrigerant expanded in the additional expansion unit **100** does not, however, bring about any cooling effect for the main mass flow **56** and must be discharged via the additional mass flow **86** and be compressed again by the additional compressor stages **70**.

As for the rest, the third embodiment of the refrigeration system according to the invention functions in a similar way to the first embodiment and so with respect to its functioning reference is also made in full to the first embodiment.

With respect to the construction of the refrigerant compressors, no further details have so far been given. Conventional reciprocating compressors can, for example, be used as refrigerant compressors.

It is particularly advantageous when, in a first preferred embodiment of such a refrigerant compressor, a cylinder head **110** as illustrated in FIGS. **7** and **8** is used and this is configured in this case for two cylinders and has an outlet chamber **112** as well as a first inlet chamber **116** and a second inlet chamber **118** which are separated from the outlet chamber **112** by a wall area **114** and, for their part, are again separated by an intermediate wall **120**.

The inlet chamber **116** is associated with one cylinder **62** of the main compressor stage **66** while the inlet chamber **118** is associated with the cylinder **62** of the additional compressor stage **70**.

For this reason, the inlet chamber **118** is also provided directly with a connection flange **122** for the additional connection **72** while the inlet chamber **116** is supplied with the refrigerant via the normal inlet channels provided in the housing.

In addition, the outlet chamber **112** is also provided with a connection flange **124** for the high pressure connection **14**.

In order to heat the refrigerant to be drawn into the inlet chambers **116** and **118** as little as possible by the refrigerant flowing into the outlet chamber **112** in a refrigerant compressor according to the invention, the wall area **114** which separates the outlet chamber **112** from the inlet chambers **116** and **118** is formed by two walls **126** and **128** which extend separately from one another over substantial areas of the height of the cylinder head **110** and between which a free space **130** is provided which insulates the walls **126** and **128** relative to one another and, therefore, also insulates the outlet chamber **112** thermally in relation to the inlet chambers **116** and **118**.

The two walls **126** and **128** are merely united essentially in a wall area **132** which borders directly on a base surface **134** of the cylinder head **110**.

The check valve **76** may preferably be arranged in the intermediate wall **120** and therefore allows refrigerant to be drawn in from the inlet chamber **116** in a simple manner when the inlet chamber **118** of the additional compressor stage **70** is not supplied with refrigerant via the additional connection **72**.

In a second, preferred embodiment of a refrigerant compressor according to the invention, illustrated in FIGS. **9** and **10**, the intermediate wall **120'** of the cylinder head **110'** is not provided with the check valve **76** but rather a check valve **176** is provided on a valve plate **140** which rests on a cylinder housing **142** and bears, for its part, the cylinder head **110'**.

For this purpose, an additional opening **144** is provided in the valve plate **140** and this opening is arranged so as to be congruent with a connecting channel **174**, which is provided in the cylinder housing **142** and branches off from the inlet channel **148**, and opens into the inlet chamber **118** for the cylinder **62** of the additional compressor stage **70**.

The opening **144** can be closed by a valve tongue **178** of the check valve **176** which is arranged on a side of the valve plate **140** facing the inlet chamber **118** and is secured, in addition, by a catcher element **180**.

The inlet chamber **116** of the main compressor stage **66** is provided with refrigerant supplied to the low pressure connection **52** via an inlet channel **148**, wherein an opening **150** is provided in the valve plate **140** which is arranged so as to be congruent with the inlet channel **148** and via which the refrigerant transfers from the inlet channel **148** into the inlet chamber **116**.

As a result, it is possible in a simple manner, as illustrated in FIG. **10**, not only to assign inlet valves, which are not, however, immediately visible in FIG. **10** and are associated with inlet openings **152** of the main compressor stage **66** and inlet openings **154** of the additional compressor stage **70**, to the valve plate **140** and, in addition, to arrange the corresponding outlet valves **156** and **158** on the valve plate **140** but also to provide the check valve **176** in the same way and preferably with the same construction as the outlet valves **156** and **158** so that this check valve can be mounted in a simple manner and can also be optimized in the same way as the outlet valves **156** and **158** with respect to its valve characteristics.

The invention claimed is:

1. Refrigeration system comprising a refrigerant circuit, a main mass flow of a refrigerant being guided in said circuit, a refrigerant-cooling heat exchanger arranged in the refrigerant circuit on the high pressure side, an expansion cooling device arranged in the refrigerant circuit, said device cooling the main mass flow of the refrigerant in the active state and thereby generating an additional mass flow of gaseous refrigerant, a reservoir for liquefied refrigerant arranged in the refrigerant circuit, at least one expansion unit for liquefied refrigerant of the main mass flow, said expansion unit being arranged in the refrigerant circuit and having an expansion element and a post-connected heat exchanger on the low pressure side, said heat exchanger making refrigerating capacity available, and at least one refrigerant compressor arranged in the refrigerant circuit, said compressor having a main compressor stage and at least one additional compressor stage driven together with the main compressor stage, said two stages compressing refrigerant to high pressure PH, wherein the main compressor stage and the at least one additional compressor stage are adapted to be used such that either the main compressor stage compresses refrigerant from the main mass flow and the additional compressor stage compresses refrigerant from the additional mass flow or the main compressor stage and the additional compressor stage compress refrigerant from the main mass flow, at least two refrigerant compressors being arranged in the refrigerant circuit, said compressors being adapted to be switched on individually for the purpose of compressing the main mass flow, at least two of the refrigerant compressors each have at least one additional compressor stage, wherein each of the additional compressor stages is adapted to be used optionally for the compression of refrigerant from the main mass flow or for the compression of refrigerant from the additional mass flow and a control provided for switching in a first operating mode as a function of the operating conditions on such a number of additional compressor stages for the compression of refrigerant from the additional mass flow that the expansion cooling device liquefies the main mass flow and reduces its enthalpy.

2. Refrigeration system as defined in claim **1**, wherein the expansion cooling device reduces the enthalpy of the main mass flow by at least 10%.

3. Refrigeration system as defined in claim **2**, wherein the expansion cooling device reduces the enthalpy of the main mass flow by at least 20%.

4. Refrigeration system as defined in claim **1**, wherein the first operating mode corresponds to a supercritical operation.

5. Refrigeration system as defined in claim **1**, wherein the expansion cooling device converts the main mass flow into a thermodynamic state, the pressure and enthalpy values of said state being lower than those of a maximum of the saturation curve.

6. Refrigeration system as defined in claim **5**, wherein the pressure and enthalpy values of the main mass flow brought about by the expansion cooling device are close to the saturation curve of the enthalpy/pressure diagram.

7. Refrigeration system as defined in claim **6**, wherein the pressure and enthalpy values of the main mass flow brought about by the expansion cooling device are essentially on the saturation curve of the enthalpy/pressure diagram.

8. Refrigeration system as defined in claim **1**, wherein the expansion cooling device has an expansion valve for the expansion of refrigerant to an intermediate pressure PZ and wherein the intermediate pressure PZ of the expansion cooling device is adjustable by switching on the suitable number of additional compressor stages.

9. Refrigeration system as defined in claim **8**, wherein the expansion valve expands refrigerant of the main mass flow and refrigerant of the additional mass flow to the intermediate pressure PZ.

10. Refrigeration system as defined in claim **8**, wherein the expansion cooling device also comprises the reservoir for the liquid refrigerant of the main mass flow.

11. Refrigeration system as defined in claim **1**, wherein in the second operating mode the expansion cooling device is in the inactive state and brings about no cooling of the main mass flow.

12. Refrigeration system as defined in claim **1**, wherein in the second operating mode all the additional compressor stages compress refrigerant of the main mass flow.

13. Refrigeration system as defined in claim **1**, wherein in the second operating mode liquid refrigerant of the main mass flow is subject to high pressure PH in the reservoir.

14. Refrigeration system as defined in claim **11**, wherein the second operating mode corresponds to a subcritical operation.

15. Refrigeration system as defined in claim **1**, wherein the control controls the refrigerant compressors in accordance with the refrigerating capacity required.

16. Refrigeration system as defined in claim **1**, wherein the refrigerant compressors are adapted to be switched on or off individually with the control in accordance with the refrigerating capacity required.

17. Refrigeration system as defined in claim **1**, wherein each refrigerant compressor with additional compressor stage is dimensioned such that the mass flow of refrigerant of the additional mass flow compressed by the additional compressor stage corresponds at the most to the mass flow of refrigerant of the main mass flow compressed by the main compressor stage in this refrigerant compressor.

18. Refrigeration system as defined in claim **1**, wherein the refrigerant compressors with additional compressor stage are dimensioned such that the additional compressor stages of different refrigerant compressors compress different mass flows of refrigerant of the additional mass flow.

19. Refrigeration system as defined in claim **1**, wherein the refrigerant compressors with additional compressor stage are reciprocating compressors.

20. Refrigeration system as defined in claim **19**, wherein each of the refrigerant compressors with additional compressor stage has at least one cylinder for the additional compressor stage and at least one cylinder for the main compressor stage.

21. Refrigeration system as defined in claim **19**, wherein the number of cylinders for the main compressor stage is greater than the number of cylinders for the additional compressor stage in each refrigerant compressor with additional compressor stage.

22. Refrigeration system comprising a refrigerant circuit, a main mass flow of a refrigerant being guided in said circuit, a refrigerant-cooling heat exchanger arranged in the refrigerant circuit on the high pressure side, an expansion cooling device arranged in the refrigerant circuit, said device cooling the main mass flow of the refrigerant in the active state and thereby generating an additional mass flow of gaseous refrigerant, a reservoir for liquefied refrigerant arranged in the refrigerant circuit, at least one expansion unit for liquefied refrigerant of the main mass flow, said expansion unit being arranged in the refrigerant circuit and having an expansion element and a post-connected heat exchanger on the low pressure side, said heat exchanger making refrigerating capacity available, and at least one refrigerant compressor arranged in the refrigerant circuit, said compressor having a main compressor stage and at least one additional compressor stage driven together with the main compressor stage, said two stages compressing refrigerant to high pressure PH, wherein the main compressor stage and the at least one additional compressor stage are adapted to be used such that either the main compressor stage compresses refrigerant from the main mass flow and the additional compressor stage compresses refrigerant from the additional mass flow or the main compressor stage and the additional compressor stage compress refrigerant from the main mass flow,

in the case of at least two refrigerant compressors with additional compressor stage the additional compressor stages of different refrigerant compressors have a different volumetric displacement.

23. Refrigeration system as defined in claim **22**, wherein for each refrigerant compressor with additional compressor stage the ratio of the volumetric displacement of the additional compressor stage to the volumetric displacement of the main compressor stage is different in relation to at least one of the other refrigerant compressors with additional compressor stage.

24. Refrigeration system as defined in claim **1**, wherein in the first operating mode the reservoir for liquefied refrigerant operates at an intermediate pressure PZ and an additional expansion unit with an expansion element and a post-connected heat exchanger delivering refrigerating capacity is provided between the heat exchanger on the high pressure side and cooling the refrigerant and the reservoir for liquefied refrigerant.

25. Refrigeration system as defined in claim **19**, wherein the refrigerant compressors have cylinder heads, outlet chambers and inlet chambers being designed in the case of the said cylinder heads so as to be essentially thermally decoupled.

26. Refrigeration system as defined in claim **1**, wherein a check valve is provided for connecting an inlet chamber of the additional compressor stage to the low pressure connection of the main compressor stage.

27. Refrigeration system as defined in claim **26**, wherein the check valve is provided in a valve plate of the respective refrigerant compressor.

28. Refrigeration system as defined in claim **27**, wherein the connecting channel between the low pressure connection and the check valve runs in a cylinder housing.

* * * * *